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Description

The present invention relates to an apparatus for controlling a variable displacement type hydraulic pump adapted to be driven by an engine.

A construction machine such as power shovel or the like is equipped with a variable displacement type hydraulic pump adapted to be driven by an engine.

A hitherto known apparatus for controlling a variable displacement type hydraulic pump has a function of properly controlling an inclinaton angle of a swash plate in the pump to assure that an output torque from the engine matches with an absorption torque absorbed by the pump at all times in order to effectively utilize the output torque from the engine.

However, the conventional apparatus has a drawback that an improvement effect covering a fuel consumption characteristic of the engine and a pump efficiency can not be expected due to the fact that the apparatus is intended to control only the variable displacement type hydraulic pump.

On the other hand, an apparatus for varying an absorption torque absorbed by a variable displacement type hydraulic pump in dependence on a given operation mode (operation to be performed under a high intensity of load, operation to be performed under a low intensity of load or the like) was already proposed by a Japanese Laid-Open Patent NO. 204987/1985.

However, the last-mentioned conventional apparatus has a drawback that it can deal with only a problem in respect of such a state that the engine is excessively heated. Incidentally, it is thinkable as a countermeasure to be taken at the time when the engine is excessively heated that an output horsepower from the engine and the number of revolutions of the engine are reduced. However, when this countermeasure with which an absorption horsepower absorbed by the pump which is a direct load exerted on the engine does not vary is employed, it not only takes a long time until a normal operational state is restored from the state that the engine is excessively heated, resulting in a satisfactory operation failing to be performed, but also a running time of the engine is shortened.

Further, the conventional apparatuses detect a pressure of hydraulic oil delivered from the pump with the use of pressure detecting means in order to control an inclination angle of a swash plate in the pump, but there arises such a problem that operation of the engine is interrupted or an output torque from the engine fails to be transmitted to the pump when an abnormality relative to the pressure detecting means occurs, because they can not entirely deal with the above-mentioned abnormality.

It is the object of the present invention to provide an apparatus for controlling a hydraulic pump which assures that a fuel consumption cost required for an engine can be reduced and moreover an operational efficiency of the hydraulic oil can be improved substantially.

This problem is solved, according to the invention, with the features of claim 1.

According to the present invention, the apparatus for controlling a variable displacement type hydraulic pump comprises means for detecting the number of revolutions of an engine, means for detecting a pressure of hydraulic oil delivered from the pump, means for setting a pump absorption torque characteristic which monotonously decreases with reference to the number of revolutions of the engine, means for looking for an inclination angle of a swash plate in the pump with reference to the pump absorption torque characteristic and the pressure of hydraulic oil delivered to the pump, means for controlling the swash plate in the pump so as to assure the aforesaid inclination angle of the swash plate, and means for reducing the number of revolution of the engine under a condition that an absorption torque absorbed by the pump does not exceed an allowable torque of the engine.

Further, according to other aspect of the present invention, an apparatus for controlling a variable displacement type hydraulic pump comprises means for detecting the number of revolutions of an engine, means for setting a target pump absorption torque T_P in accordance with the following equation,

$$T_P = T_E (N_c) + K(N - N_c)$$

where

$T_E (N_c)$;	rated torque characteristic of the engine at a predetermined revolution number range
K ;	constant
N ;	number of revolutions of the engine
N_c ;	target number of revolutions of the engine

and means for controlling a swash plate in the pump so as to obtain an absorption torque with reference to the target pump absorption torque and a pressure of hydraulic oil delivered from the pump.

The apparatus for controlling a variable displacement type hydraulic pump comprises means for setting a pump absorption torque characteristic so as to reduce an absorption torque absorbed by the pump lower

than an output torque from the engine and means for controlling an inclination angle of a swash plate in the pump so as to allow the absorption torque absorbed by the pump to exhibit a value which conforms to the pump absorption torque characteristic when means for detecting a pressure of hydraulic oil delivered from the pump becomes abnormal in function.

The apparatus for controlling a hydraulic pump assures that the pump can be operated even at the time when means for detecting a pressure of hydraulic oil delivered from the pump becomes abnormal in function.

Fig. 1 is a block diagram illustrating an apparatus for controlling a hydraulic pump in accordance with an embodiment of the present invention, Fig. 2 is a flow chart illustrating procedures for a controller, Fig. 3 is a graph illustrating a function of the apparatus shown in Fig. 1, Fig. 4 is a schematic view of a proportion solenoid for actuating a fuel control lever, Fig. 5 is a graph exemplifying pump absorption torque characteristics corresponding to a magntitude of work to be undertaken, Fig. 6 is a graph exemplifying a relationship between an inclination angle of a swash plate and a torque efficiency, Fig. 7 is a graph exemplifying a relationship between the number of revolutions of an engine and a fuel consumption cost, Fig. 8 is a block diagram illustrating an apparatus for controlling a hydraulic pump in accordance with other embodiment of the present invention, Fig. 9 is a block diagram exemplifying a structure of a controller shown in Fig. 8, Fig. 10 is a graph exemplifying an output horsepower characteristic of an engine, Fig. 11 is a graph illustrating a relationship between a torque characteristic of an engine and an absorption torque of a hydraulic pump, Fig. 12 is a graph illustrating an output characteristic of a function generator, Fig. 13 is a block diagram illustrating an apparatus for controlling a hydraulic pump in accordance with another embodiment of the present invention, Fig. 14 is a flow chart exemplifying processing procedures of a controller shown in Fig. 13, Figs. 15 and 16 are a graph exemplifying a relationship between a horsepower generated by an engine and a horsepower absorbed by a hydraulic pump respectively, Fig. 17 is a flow chart illustrating processing procedures of a controller at the time when a pressure sensor becomes abnormal in function, Figs. 18 and 19 are a graph exemplifying a relationship between a rated torque of an engine and an absorption torque characteristic of a hydraulic pump applicable at the time when the pressure sensor becomes in function, respectively, and Fig. 20 is a graph showing a magnitude of absorption torque in a case where the pump absorption torque characteristic shown in Fig. 19 is applied.

Now, the present invention will be described in a greater detail hereunder with reference to the accompanying drawings which illustrate preferred embodiments thereof.

As is apparent from Fig. 6, a hydraulic pump has an advantage in terms of torque efficiency when it is operated with a high magnitude of inclination angle of a swash plate. Further, the hydraulic pump has an advantage in terms of reduction of fuel consumption cost when an engine is operated with a number of revolutions thereof which is reduced to a certain level, as shown in Fig. 7.

Refering to Fig. 1 which schematically illustrates an apparatus for controlling a variable displacement type hydraulic pump in accordance with an embodiment of the invention, the following relationship is established when an absorpsion horsepower absorbed by the variable displacement type hydraulic pump 2 driven by an engine 1 is represented by W_p.

$$\begin{aligned} W_p &= K_1 \cdot P \cdot Q \\ &= K_2 \cdot P \cdot N \cdot V \qquad \text{---} \quad (1) \end{aligned}$$

where

P ;	pressure of hydraulic oil delivered from the pump	(Kg/cm^2)
Q ;	flow rate of hydraulic oil delivered from the pump	(liter/min)
N ;	number of revolutions of the engine	(rpm)
V ;	flow rate of hydraulic oil delivered from the pump per one revolution of the pump	(cc/rev)
K_1, K_2 ;	constant	

As will be readily understood from the above Equation (1), Q (N • V) is determined by N and V, and each of these parameters can take various values. Namely, to obtain a same value of Q, it suffices that a value of N is decreased and a value of V is increased correspondingly. For instance, by properly controlling a value of Q in relation to a voluntary value of P, the absorption horsepower W_P absorbed by the pump 2 can be so controlled that it is kept constant.

A pump absorption torque $T_{P\text{-}W}$ required for controlling in order that the absorption horsepower W_P absorbed by the pump 2 is kept constant is represented by the following equation.

$T_{P\text{-}W} = K_3 \cdot \frac{W}{N} = f(N)$ (2)

where

W ; constant work to be conducted by the pump

K_3 ; constant

Further, to obtain the absorption torque $T_{P\text{-}W}$, a flow rate V of hydraulic oil delivered from the pump 2 per one revolution of the pump 2 is represented by the following equation.

$$V = \frac{T_{P\text{-}W}}{K_4 \cdot P} = \frac{f(N)}{K_4 \cdot P} \quad \cdots (3)$$

where

K_4 ; constant

Accordingly, a torque efficiency of the pump can be increased and a fuel consumption cost of the engine 1 can be reduced under such a condition that the absorption horsepower W_P absorbed by the pump is maintained at a constant value of W, if the engine is controlled so as to reduce N on the assumption that the absorption torque $T_{P\text{-}W}$ absorbed by the pump is represented as a monotonously decreasing function A (hyperbolic function) using the number N of revolutions of the engine as a variable as shown in Fig. 3 and V is represented as a function which is obtained by dividing f (N) by P.

It should be noted that since V has the maximum value V_{max} which is set under a rated condition of the pump 2, N can not be reduced thoughtlessly. Further, as is apparent from the Equation (2), since the absorption torque $T_{P\text{-}W}$ increases as N is reduced, there is a danger that the absorption torque $T_{P\text{-}W}$ exceeds a rated torque B shown in Fig. 3 in dependence on an extent of reduction of N. Accordingly, in view of the above-mentioned fact, N can be reduced thoughtlessly. Namely, as shown in Fig. 3, the number of revolutions of the engine can not be reduced lower than N_L, because the absorption torque $T_{P\text{-}W}$ absorbed by the pump is in excess of the rated torque of the engine in a case where the number of revolutions of the engine is reduced lower than N_L.

In an embodiment of the present invention to be described below, improvement of an operational efficiency of the pump as well as improvement of fuel consumption cost are achieved while while the above-mentioned facts are taken into account.

Incidentally, the aforesaid rated torque B is set by means of a governor 10. Pressurized hydraulic oil delivered from the pump 2 is fed to a hydraulic actuator (hydraulic motor, hydraulic cylinder or the like) usable for a construction machine which is not shown in the drawings.

Refering to Fig. 1 again, a signal corresponding to an extent of actuation of an acceleration lever 4 is outputted from an acceleration sensor 3, a signal representative of the actual number N of revolutions of the engine 1 is outputted from an engine rotation sensor 5, and a signal representative of a pressure P of hydraulic oil delivered from the pump 2 is outputted from a pressure sensor 6. Each of the output signals outputted from these sensors is inputted into a controller 7.

The signal outputted from the acceleration sensor 3 is subjected to amplifying or the like processing in the controller 7 and thereafter it is inputted as a signal representative of the target number N_r of revolutions of the engine into a proportion solenoid 9 which will be described later.

The actuator 8 for driving a swash plate is composed of, for instance, a servo valve, a hydraulic cylinder and others each of which is not shown in the drawings, and a swash plate 2a in the pump 2 is driven by the actuator 8.

A pump absorption torque characteristic A and the number N_L of revolutions of the engine both of which are shown in Fig. 3 are previously stored in a memory 12.

As shown in Fig. 4, the proportion solenoid 9 is provided as an actuator for actuating a fuel control lever 11 on the governor 10 and an amount of fuel injection varies in dependence on an extent of displacement of the control lever 11 achieved under the effect of actuating force of the proportion solenoid 9.

Each of a plurality of regulation lines ℓ_1, ℓ_2 and others as shown in Fig. 3 is set in dependence on a magnitude of the target number N_r of revolutions of the engine and, for instance, the regulation line set in a case where the acceleration lever 4 is turned to a full throttle position is identified by ℓ_1.

Now, when it is assumed that the acceleration lever 4 is turned to the full throttle position and the variable displacement type hydraulic pump 2 is conducting a work W, a torque developed at an intersection P_1 where the regulation line ℓ_1 intersects the pump absorption torque characteristic A represents a

matching torque for both the engine 1 and the pump 2, and the number of revolutions of the engine measured at this moment is identified by N_1.

According to the embodiment of the present invention, the number of revolutions of the engine is caused to decrease from the state that the acceleration lever 4 is turned to the full throttle position. Now, the embodiment of the present invention will be described below in more details with reference to Fig. 2 which shows a plurality of processing procedures in the controller 7.

In the controller 7, the number N of revolutions of the engine and a pressure P of hydraulic oil delivered from the pump 2 are first detected in response to an output from the engine rotation sensor 5 and the pressure sensor 6 (Step 100) and the pump absorption torque $T_{P\cdot W}$ represented by the Equation (2) and corresponding to the detected number N of revolutions of the engine is then read out of the memory 12 with reference to the detected number N of revolutions of the engine (Step 101).

Next, an arithmetic operation represented by the Equation (3) is executed with reference to the read absorption torque $T_{P\cdot W}$ and the pressure P of hydraulic oil from the pump detected during the Step 100 (Step 102) and thereby a Flow rate V of hydraulic oil delivered from the pump 2 per one revolution thereof is obtained. Incidentally, due to the fact that V and an inclination angle of the swash plate have a corresponding relationship therebetween as represented by a ratio of 1 : 1, the result is that the arithmetic operation executed during the Step 102 is intended to obtain an inclination angle of the swash plate.

Next, a command relative to the inclination angle for obtaining a flow rate V of hydraulic oil from the pump detected during the Step 102 is prepared and it is then applied to the actuator 8 for driving the swash plate (Step 103) whereby the absorption torque $T_{P\cdot W}$ of the pump 2 represents a value at the point P_1 in Fig. 3.

During next Steps 104 and 105, a processing for comparing V obtained during the Step 102 with threshold values V_{M1} and V_{M2} is executed. The threshold values V_{M1} and V_{M2} are set to, for instance, 90 % and 80 % of the maximum value V_{max} of V which is determined under a rated condition of the pump 2, and it is judged by them whether or not the swash plate in the pump 2 is driven to an angular position located in the proximity of the maximum inclination angle.

Now, when it is assumed that results of the comparison made during the Steps 104 and 105 are represented by an inequality of $V < V_{M2}$, that is to say, the swash plate in the pump 2 is not driven to an angular position in the proximity of the maximum inclination angle, a time-up equal to time Δt_1 (for instance, 100 ms) is judged by means of a first timer incorporated in the controller 7 (Step 106) and thereafter a comparison is made between the preset limitative number N_L of revolutions of the engine (see Fig. 3) stored in the memory 12 and the existent number N (= N_1) of revolutions of the engine (Step 107).

Since an inequality of $N > N_L$ is established at this moment, a processing for reducing the number of revolutions of the engine from the existent number of revolutions of the engine by an extent of Δ N (for instance, 15 rpm) is executed in the controller 7 (Step 108). That is to say, a proceeding for changing to $N_r - \Delta N$ the target number N_r of revolutions of the engine commanded by actuation of the lever 4 is executed whereby the proportion solenoid 9 is actuated so as to reduce the number of revolutions of the engine 1 by an amount of Δ N.

Thereafter, as long as results of the comparison made during the Step 105 is represented by an inequality of $V < V_{M2}$ and results of the comparison made during the Step 107 are represented by an inequality of $N > N_L$, procedures shown in the Steps 100 to 108 are executed repeatedly. That is to say, the target number of revolutions of the engine is changed in accordance with the following manner

$N_r \rightarrow (N_r - \Delta N) \rightarrow (N_r - \Delta 2 N) \rightarrow (Nr - \Delta 3 N)$ ---

whereby the number of revolutions of the engine is reduced by a step of Δ N. As the number of revolutions of the engine is reduced in the above-described manner, the absorption torque $T_{P\cdot W}$ read out of the memory 12 becomes larger, as shown by the characteristic A in Fig. 3, and thereby a value of command relative to an inclination angle to be outputted during the Step 103 becomes larger correspondingly. That is to say, an inclination angle of the swash plate in the pump 2 is increased.

Changing of the aforesaid target number of revolutions of the engine means that the regulation lines as shown in Fig. 3 are set in accordance with the following manner $\ell_1 - \ell_2 - \ell_3$ - ---. Thus,the matching point relative to torque is changed in accordance with the following manner

$P_1 \rightarrow P_2 \rightarrow P_3 \rightarrow$ ---.

While the number of revolutions of the engine is reduced in the above-described manner, a proceeding for reducing the number N of revolutions of the engine is interrupted when it is judged during the Steps 104 and 105 that an inequality of $V_{M1} \geq V \geq V_{M2}$ is established, and thereby procedures are caused to return to the Step 100.

Further, in a case where P is changed to decrease in accordance with variation of load (to reduce a load to be exerted on the pump) and it is then judged during the Step 104 that an inequality of $V > V_{M1}$ is

established, a processing for increasing the existent number of revolutions of the engine by an amount of Δ N is executed (Step 110) after a time-up equal to Δ t_2 is judged by a second timer (Step 109).

Since the processing executed during the Step 110 reduces $T_{P.W}$ shown in the Step 101, the result is that an inclination angle of the swash plate in the pump becomes smaller.

Next, description is made below as to a case where it is continuously judged during the Step 105 that an inequality of $V < V_{M2}$ is established and it is judged during the Step 107 that an equality of $N = N_L$ is established. In this case, since the absorption torque $T_{P.W}$ absorbed by the pump 2 is in excess of a torque allowable for the engine 1 as the number N of revolutions of the engine is reduced lower than the above-mentioned level, a processing to be executed during the Step 108 fails to be executed and thereby the procedures are caused to return to the Step 100 irrespective of the state that the existent inclination angle is smaller than the inclination angle corresponding to the threashold value V_{M2}.

As will be apparent from the above description, the number N of revolutions of the engine is reduced as far as possible and an inclination angle of the pump is increased in accordance with this embodiment of the invention. Consequently, it follows that the pump 2 can be operated under a condition of high torque efficiency and the engine 1 can be operated in a rotational range where a low fuel consumption rate is assured.

Refering to Fig. 3 again, merely the characteristic A relative to an absorption torque in a case where the pump 2 is adapted to absorb a constant horse power W is shown in the drawing but, in practice, a plurality of characteristics relative to an absorption torque corresponding to a magnitude of absorption horsepower are set. For instance, absorption torque characteristics A_1 and A_2 corresponding to absorption horsepowers W_{P1} and W_{P2} are set as shown in Fig. 5 and they are stored in the memory 12. A mode for selecting a work W_1 is selected when a light work is undertaken, whereas a mode for selecting work W_2 is selected with the use of an operation mode shifting switch 13 shown in Fig. 1 when a heavy work is undertaken. Thus, the characteristic A_1 or A_2 is designated by such an operation for selecting a certain mode as mentioned above.

In the above-described embodiment, the absorption torque characteristic A represents a hyperbolic function as identified by an equation of $f(N) = W/N$. However, a monotonously decreasing function approximate to the above-noted function f (N), for instance, a function as represented by a dotted line in Fig. 5 which varies in inverse proportion to an increase of the number N of revolutions of the engine may be employed as a function representative of the characteristic A. In this case, it should of cource be understood that a relation which represents that a value of $W \cdot P \cdot Q$ is kept constant collapses to some extent as the number of revolutions of the engine varies. However, in some case, it will be preferable to carry out such controlling as mentioned above in dependence on an intensity of load.

Incidentally, in the above-described embodiment, controlling is achieved for N and V in order that a product of n_E multiplied by n_P reaches the maximum value, when it is assumed that a fuel consumption rate of the engine 1 is represented by a function of $n_E = F(N)$ relative to N and an operational efficiency of the pump 2 corresponding to an inclination angle of the swash plate is represented by a function of $n_P = G(V)$ relative to V.

Fig. 8 illustrates other embodiment of the present invention.

Refering to the drawing, an engine 21 has a rated horse power characteristic as shown in Fig. 10. That is to say, it has a horsepower characteristic which assures that it can obtain a constant horsepower in a range as defined between number N_b of revolution of the engine and number N_a of revolutions of the engine. Fig. 11 illustrates a rated torque characteristic C for obtaining the above-noted rated horsepower characteristic and this torque characteristic is set with the aid of a governor (not shown) attached to the engine 22.

The number N of revolutions of the engine is detected by means of an engine rotation sensor 23 and an inclination angle 0 of the swash plate in a pump 22 is detected by means of an angle sensor 24.

A torque command to be issued to the pump 22 and a pressure of hydraulic oil delivered from the pump 22 are inputted into a variable regulator 25, and a swash plate 22a in the pump 22 is driven in such a manner that the pump 22 absorbs a torque in response to the torque command.

As shown in Fig. 9, a controller 26 is composed of a revolution number command generating section 260 for commanding a target number N_c of revolutions of the engine, a limiter 261 for limiting the number N_c of revolutions of the engine between the maximun value $N_{c\ max}$ (corresponding to N_a) and the minimum value $N_{c\ min}$ (corresponding to N_b), a function generator 262 for generating a command torque T_a corresponding to the number N_c of revolutions of the engine in response to an output from the command generating section 260, a comparator 263 for comparing the inclination angle 0 of the swash plate detected by means of the angle sensor 24 with the maximum value of θ_{max} to generate a reduction command DN of the command revolution number N_c when an inequality of $\theta < \theta_{max}$ is established, a substractor 264 for

obtaining a deviation ($N - N_c$) of the number N of revolutions of the engine from the command number N_c of revolutions of the engine, a comparator 265 adapted to output an increase command UP relative to N_c when the deviation ($N - N_c$) becomes larger than a preset value SD, an amplifier 266 For amplifying the deviation ($N - N_c$) by K times, and an adder 267 for adding the command torque T_a to the deviation $K(N - N_c)$ amplified by K times.

The revolution number command generating section 260 functions for reducing N_c by number of revolutions identified by ΔN_c at a predetermined time interval when a reduction command DN is outputted from the comparator 263 and increasing N_c by number of revolutions identified by ΔN_c at the predetermined time interval when an increase command UP is outputted from the comparator 265.

The function generator 262 has a variation pattern as shown in Fig. 12 corresponding to a variation pattern as seen in a range from N_a to N_b relative to a rated torque characterisstic C shown in Fig. 11. This causes a command torque $T_E(N_c)$ generated in the function generator 262 to become a function which varies in dependence on the command revolution number N_c.

The revolution deviation $K(N - N_c)$ amplified by K times in the amplifier 266 is a primary function relative to the inclination K and is caused to move in parallel in accordance with variation of N_c.

A function represented by the following Equation (4) to which functions $T_E(N_c)$ and $K(N - N_c)$ relative to the command torque are added is obtainable in the adder 267.

$$T_P = T_E(N_c) + {}_K(N - N_c) \qquad (4)$$

The function of the above Equation (4) is represented by lines D, E and F shown by dotted lines in Fig. 11 when N_c assumes $N_{c\,max}$, $N_{c\,mid}$ and $N_{c\,min}$.

In a case where the absorption torque T_p of the pump 22 is varied in accordance with the function of the Equation (4), the absorption torque T_p matches with the rated torque of the engine 21 at a point P_a shown in Fig. 11, for instance, when N_c assumes $N_{c\,max}$.

Next, operation of the apparatus in accordance with this embodiment will be described below.

In the revolution number command generating section 260 shown in Fig. 9, for instance, the number of revolutions of the engine as identified by $N_c = N_{c\,max}$ is commanded at the early part of operation. At this moment, when it is assumed that the inclination angle θ of the swash plate in the pump 22 is represented by $\theta < \theta_{max}$ in the comparator 263, a reduction command DN relative to the command number N_c of revolutions of the engine is outputted from the comparator 263. As a result, a processing for reducing the command number N_c of revolutions of the engine by number of revolutions as identified by ΔN_c (for instance, 15 to 20 rpm) at a time interval identified by time ΔT (for instance, 100 ms) in the revolution number command generation section 260 is executed. Since the command relative to the number N_c of revolutions of the engine is issued also to a governor (not shown) on the engine 21, it follows that the number of revolutions of the engine 21 is reduced by a step of ΔN_c at every time when the above-mentioned processing is executed.

On the other hand, a command signal indicative of the torque T_p represented by the Equation (4) is outputted from the adder 267 shown in Fig. 9 so that it is applied to the variable regulator 25. The variable regulator 25 drives the swash plate 22a in accordance with a relation as represented by the command torque T_p, a pressure P of hydraulic oil delivered from the pump 22 and the following Equation (5) in order that an absorption torque of the pump 22 becomes the command torque T_p.

$$V = K_5 \frac{T_P}{P} \qquad \cdots (5)$$

where

V ; volume of hydraulic oil discharged from the pump per one revolution thereof

K_5 ; constant

V in the above Equation (5) corresponds to an inclination angle θ of the swash plate, and the variable regulator 25 functions for varying the inclination angle 0 of the swash plate so as to obtain V.

When the number N of revolutions of the engine is varied by a step of ΔN_c in the above-described manner, the pump load line D shown in Fig. 11 is caused to move toward another line F. This means that V in the Equation (5) is increased, that is to say, the inclination angle 0 of the swash plate becomes larger.

When the inclination angle θ of the swash plate is increased to reach an angular position represented by $\theta = \theta_{max}$, the revolution number reduction command DN to be issued from the comparator 263 is

interrupted.

Thus, according to this embodiment, the number of revolutions of the engine can be reduced as far as possible under such a condition that the engine is operated with a constant horsepower, and an inclination angle of the swash plate in the pump can be enlarged. Accordingly, an advantageous effect that a fuel consumption cost can be reduced and the pump can be operated at a high operational efficiency is obtained in the same manner as in the preceding embodiment.

Incidentally, in the preceding embodiment, the above-mentioned advantageous effect is obtained while the pump is operated with a constant horsepower, whereas in the embodiment as shown in Fig. 8, the advantageous effect is obtainable while the engine is operated with a constant horse power.

In a case where, for instance, an operator performs an operation for reducing load exerted on the pump 22 while the latter is operated under the condition of $\theta = \theta_{max}$, a difference $(N - N_c)$ in number of revolutions becomes larger as the number N of revolutions of the engine increases. Incidentally, the difference $(N - N_c)$ in number of revolutions usually exhibits a value of substantially zero.

The comparator 265 shown in Fig. 9 is adapted to add a revolution number increase command UP to the revolution number command generating section 260, when $(N - N_c)$ is in excess of a preset value SD, that is to say, when a load exerted on the pump 22 is reduced lower than a predetermined value.

As a result, a command number N_c of revolutions of the engine is increased by number of revlutions identified by ΔN_c at a time interval as identified by ΔT, and a processing for increasing the target number of revolutions of the engine continues until a difference $(N - N_c)$ in number of revolutions becomes smaller than a value of SD, that is to say, until a load torque (pump absorption torque absorbed by the pump) matches with an engine torque.

Thus, according to this embodiment, when a load exerted on the pump 22 is reduced rapidly, N_c is caused to automatically increase and a matching point where the pump absorption torque absorbed by the pump matches with the engine torque is varied until a difference $(N - N_c)$ in number of revolutions becomes substantially zero.

Incidentally, in the foregoing embodiment, it is naturally possible to assure functions of the controller 26 shown in Fig. 9 with the aid of program controlling to be effected by a microcomputor.

Further, in the foregoing embodiment, a target inclination angle of the swash plate is mechanically obtained by introducing into the variable regulator 25 a pressure P of hydraulic oil delivered from the pump 22. However, the present invention should not be limited only to this. Alternatively, the target inclination angle of the swash plate may be electrically obtained by electrically detecting the pressure P of hydraulic oil delivered from the pump by means of a pressure sensor and utilizing an output from the pressure sensor as well as an output from the adder 267.

Further, in the embodiment, an actual inclinatiuon angle θ of the swash plate is detected by means of the angle sensor 24 shown in Fig. 8 and it is then added to the comparator 263. However, it is naturally possible to use the aforesaid electrically obtained target inclination angle in place of the acutal inclination angle θ which is obtained by means of the angle sensor 24.

Fig. 13 illustrates another embodiment of the present invention which is intended to deal with a problem in relation to overheating of the engine.

Incidentally, in the drawing, an engine 31, a pump 32, an acceleration sensor 33, an acceleration lever 34, an engine rotation sensor 35, a pressure sensor 36, an actuator 38 for driving a swash plate, a proportion solenoid 39 and a governor 40 are in common with those shown in Fig. 1 and therefore their repeated description will not be required.

A temperature sensor 41 serving as overheat detecting means outputs a signal indicative of a temperature T of the engine 31 (for instance, temperature of cooling water, temperature of exhaust gas or the like). Further, an operation mode shifting switch 42 is actuated by an operator in dependence on the operating condition, and a H mode for operation with a high intensity of load, a M mode for operation with an intermediate intensity of load and a L mode for operation with a low intensity of load are selectively indicated by the switch 42.

Now, when it is assumed that a generation horsepower generated by the engine 31 is identified by W_E and an absorption horse power absorbed by the hydraulic pump 32 is identified by W_P, they are represented in the following manner under a certain load condition.

$$W_E = W_P = K_1 \cdot P \cdot Q = K_2 \cdot P \cdot N \cdot V \quad (6)$$

where

P ;	pressure of hydraulic oil delivered from the pump	(Kg/cm^2)
Q ;	flow rate of hydraulic oil delivered from the pump	(liter/min)

V ; flow rate of hydraulic oil delivered from the pump per 30 revolutions thereof (cc/rev)
K_1, K_2 ; constant

And, the following relationship is obtained from the Equation (6).

$$V = \frac{W_P}{K_2 \cdot P \cdot N} \quad \cdots (7)$$

Incidentally, as already mentioned above, V corresponds to an inclination angle of the swash plate 32a in a ratio as represented by 1 : 1. Accordingly, V in the Equation (7) suggests an inclination angle of the swash plate.

In Fig. 15 reference character R designates a rated horse power characteristic of the engine 31, that is to say, it does a horsepower characteristic under a condition that the acceleration lever 34 is actuated to a full position.

Usually, a construction machine is operated under a condition that the acceleration lever 34 is actuated to the full position and at this moment the maximum horsepower point of the engine 31 is represented by P_1.

Lines G_1, G_2 and G_3 shown in the drawing represent an absorption horsepower characteristic of the pump respectively which is set previously. These horse power characteristics represent monotonously increasing functions f_1 (N), f_2 (N) and f_3 (N) with respect to the number N of revolutions of the engine and they intersect a rated horsepower characteristic R of the engine 31 at points P_1, P_2 and P_3.

These horsepower characteristics are previously stored in the memory 43 shown in Fig. 13.

In order to vary an absorption horsepower W_P absorbed by the pump 32 represented in the Equation (7) in accordance with the functions f_1 (N), f_2 (N) and f_3 (N), it suffices that an inclination angle of the swash plate in the pump 32 is controlled so as to obtain V as represented by the following Equations (8), (9) and (10).

$$V = \frac{f_1 (N)}{K_2 \cdot P \cdot N} \quad \cdots (8)$$

$$V = \frac{f_2 (N)}{K_2 \cdot P \cdot N} \quad \cdots (9)$$

$$V = \frac{f_3 (N)}{K_2 \cdot P \cdot N} \quad \cdots (10)$$

When an inclination angle of the swash plate in the pump 32 is controlled in accordance with the Equations (8), (9) and (10) under a condition that the throttle lever 34 is actuated to a full position, it follows that the generation horsepower W_E generated by the engine 31 matches with the absorption horsepower W_P absorbed by the pump 32 at the points P_1, P_2 and P_3.

Further, when an amount of actuation of the throttle lever 34 is reduced and thereby the number of revolutions of the engine is reduced by an amount of Δ N, that is to say, when a horsepower characteristic of the engine 31 is set as represented by a reference character R' in Fig. 15, it follows that the generation horsepower W_E generated by the engine 31 matches with the absorption horsepower W_P absorbed by the pump 32 at the points P_1', P_2' and P_3' by controlling an inclination angle of the swash plate in accordance with the Equations (8), (9) and (10).

Fig. 14 illustrates processing means for a controller 44 shown in Fig. 13.

With respect to procedures to be executed, it is first judged whether or not an operation mode L is indicated by means of the operation shifting switch 42 (Step 200), and when it is found that the operation mode L is not indicated, it is judged during a next Step 201 whether an operation mode M is indicated or not. When it is found that both the operation modes L and M are not indicated, that is to say, when an operation mode H is indicated, it is judged during a next Step 203 whether the engine 31 is excessively heated or not, and when it is found that the result of judgement is represented by NO, among absorption

horsepower characteristics G_1, G_2 and G_3 in Fig. 15 stored in the memory 43, the characteristic $G_1 = f_1$ (N) is selected (Step 208).

On the other hand, when the result of judgement made during the Step 201 is represented by YES, it is judged during a Step 209 whether the engine 31 is excessively heated or not, and when it is fount that the engine 31 is not excessively heated, the characteristic $G_2 = f_2$ (N) shown in Fig. 15 is selected during a Step 204. Further, when the result of judgement made during the Step 200 is represented by YES, the characteristic $G_3 = f_3$ (N) shown in the drawing is selected during a Step 211.

It should be noted that judgement to be made during the Steps 202 and 209 as to whether the engine is excessively heated or not is made in response to an output from the temperature sensor 41.

After a processing for making a slection during either of the Steps 208, 204 and 211 is executed, the number N of revolutions of the engine 31 is detected in response to an output from the engine rotation sensor 35 and a pressure P of hydraulic oil delivered from the pump 31 is detected in response to an output from the pressure sensor 36 (Step 205).

When the characteristic $G_1 = f_1$ (N) is selected during the Step 208, the arithmetic operation as represented in the Eqation (8) is executed during a Step 206 with reference to the characteristic f_1 (N) and N and P detected during the Step 205 whereby a flow rate V of hydraulic oil delivered from the pump is obtained in order that the absorption horse power W_P absorbed by the pump 32 assumes a value which conforms to f_1 (N).

Further, when the characteristic $G_2 = f_2$ (N) is selected during the Step 204 and the characteristic $G_3 = f_3$ (N) is selected during the Step 211, the arithmetic operations shown in the Equtaions (9) and (10) are executed during the Step 206 whereby a flow rate V of hydraulic oil delivered from the pump is obtained in order that the absorption horsepower W_P absorbed by the pump assumes values which conforms to f_2 (N) and f_3 (N).

A swash plate inclination angle command (which is represented by a value corresponding to V) for obtaining a flow rate V of hydraulic oil from the pumnp detected during the Step 206 is prepared during a next Step 207 and it is then outputted to the actuator 38 for driving the swash plate.

As a result, the acceleration lever 34 is set to a full position, and in a case where it is found that the engine 31 is not excessively heated, it follows that an absorption horsepower absorbed by the pump 32 matches with a generation horsepower generated by the engine 31 at the points P_1, P_2 and P_3 shown in Fig. 15, when the characteristic $G_1 = f_1$ (N), and the characteristic $G_2 = f_2$ (N) and the characteristic $G_3 = f_3$ (N) are selected.

That is to say, in a case where the mode H is selected and operation is performed with a high intensity of load, a horsepower at the point P_1 is absorbed by the pump 32. Further, in a case where the mode M is selected and operation is performed with an intermedtate intensity of load as well as in a case where the mode L is selected and operation is performed with a low intensity of load, horsepower at the points P_2 and P_3 are absorbed by the pump 32.

When operation is performed in accordance with the mode H or the mode L, in some case, the engine 31 is excessively heated due to an increased load.

According to the procedures shown in Fig. 14, in a case where it is judged during the Step 202 that the engine is excessively heated when the mode H is indicated, a processing for reducing the number of revolutions of the engine by Δ N is executed durin the Step 203, and the absorption horse power characteristic $G_2 = f_2$ (N) is selected during the Step 204.

Further, in a case where it is judged during the Step 209 that the engine is excessively heated when the mode M is indicated, similarly a processing for reducing the number of revolutions by Δ N is executed during the Step 210, and the absorption horsepower characteristic $G_3 = f_3$ (N) is selected during the Step 211.

Incidentally, a processing to be executed during the Step 203 or 210 means that a signal indicative of the target number N_r of revolutions of the engine applied to the proprotion solenoid 39 is changed to a signal indicative of the number $N_r - \Delta$ N of revolutions of the engine. Thus, a horsepower characteristic of the engine 31 is represented by R' in Fig. 15.

Thereafter, the above-mentioned processings are executed during the Steps 205, 206 and 207. Thus, in a case where the engine is excessively heated under a condition that the mode H is indicated, a matching point where the absorption horsepower W_P absorbed by the pump 32 matches with the generation horsepower W_E generated by the engine 31 is shifted from the point P_1 to the point P_2' in Fig. 15. Further, in a case where the engine is excessively heated under a condition that the mode M is indicated, the matching point is shifted from the point P_2 to the point P_3'.

It should be noted that processings to be executed during the Steps 203 and 210 for the purpose of reducing the target number of revoltuions of the engine by Δ N continue until the excessively heated state

of the engine disappears.

When the matching point is shifted from the point P_1 to the point P_2' or it is shifted from the point P_2 to the point P_3', a load exerted on the engine 31 is reduced remarkably. Accordingly, the engine 31 can be quickly restored to the normal operative state from the excessively heated state. Since a controlling operation for the swash plate in the pump 31 continues while the above-mentioned processings are executed, a work can proceed further without any occurrence of malfunction such as remarkable reduction of the number of revolutions of the engine or the like.

Incidentally, in this embodiment, the characteristics G_1, G_2 and G_3 shown in Fig. 15 are stored in the memory 43. However, it is possible to allow the controller 44 to arithmetically process pump absorption horse powers which conform to these characteristics.

Further, in the above-described embodiment, a practical manner to be employed when the acceleration lever 34 is actuated to a full position is shown. However, it should be noted that even in a case where the lever 34 is actuated to an intermediate operation position, it is possible to effect controlling in the same manner as mentioned above. In this case, it should of cource be understood that also the characteristics f_1 (N), f_2 (N) and f_3 (N) in relation to the intermediate position are stored in the memory 43.

Further, in this embodiment, each of the pump absorption horsepower characteristics $G_2 = F_2$ (N) and $G_3 = f_3$ (N) is represented in the form of a monotonously increasing function relative to the number N of revolutions of the engine. As shown in Fig. 16, however, a constant function (constant horsepower characteristic) relative to N may be practically employed for these characteristics.

In a case where an absorption torque absorbed by the pump is controlled in accordance with the characteristic A shown in Fig. 3, the characteristics D, E and F shown in Fig. 11 or the characteristics G_1, G_2 and G_3 shown in Fig. 15, it is necessary to detect a pressure of hydraulic oil discharged from the pump. Conversely speaking, when it becomes impossible to detect a pressure of hydraulic oil discharged from the pump, it follows that the above-mentioned torque controlling failes to be effected properly, resulting in an occurrence of malfunction such as interruption of operation of the engine due to excessive load, complete failure of transmission of a torque outputted from the engine or the like.

Fig. 17 illustrates procedures for avoiding an occurrence of the above-mentioned malfunction, and the procedures are executed by means of the controller 7 shown in Fig. 1 or the controller 44 shown in Fig. 13.

The hydraulic pump 2 or 32 has the maximum delivery pressure P_{max} which can be outputted. Accordingly, when a pump absorption torque characteristic T_P (N) which is not in excess of a rated torque of the engine, for instance, as shown by a dotted chain line in Fig. 18 is previously set and a flow rate V of hydraulic oil delivered from the pump per one revolution thereof is controlled so as to satisfactorily meet a relation as represented by the following equation, an absorption torque absorbed by the pump does not exceed an output torque I from the engine 2.

$$V = \frac{T_P\ (N)}{K \cdot P_{max}} \qquad \cdots (11)$$

where

K ; constant

Here, the controllers 7 and 44 are so constructed that the limitative torque characteristic T_P (N) and the maximum delivery pressure P_{max} are previously stored in the memory.

Incidentally, the limitative torque characteristic T_P (N) is set so as to obtain an absorption torque as large as possible on the assumption that operation of the engine is not interrupted.

According to the procedures shown in Fig. 17, it is first judged whether or not there is existent an abnormality with the pressure sensors 6 and 36 (Step 300). Incidentally, this judgement is made, for instance, in the following manner. Namely, when the sensors 6 and 36 have a pressure detection range of 0 to 50 Kg/cm^2, their output voltage varies, for instance, in the range of 1 to 5 V in dependence on variation of the pressure P. Thus, when it is found that the output voltage is not in the range 1 to 5 V, it is judged by means of the controllers 7 and 44 that the sensors 6 and 36 are abnormal in function.

When it is judged during the Step 300 that the pressure sensor is abnormal in function, the number N of revolutions of the engine is inputted (Step 301), and an arithmetic operation shown in Equation (11) is then executed with reference to the number N of revolutions of the engine, the limitative torque characteristic T_P (N)) shown in Fig. 18 and the maximum delivery pressure P_{max} whereby a target flow rate V of hydurlic oil delivered from the pump is obtained. And, a swash plate inclination angle command for obtaining V is prepared and it is then outputted to the actuator 8 or 38 (Step 303) whereby an absorption

torque to be absorbed by the pump is controlled in accordance with the torque characteristic T_P (N).

Incodentally, in a case where it is not detected during the Step 300 that the pump is abnormal in function, normal torque controlling is executed with reference to an output from the pressure sensor (Step 304).

In the foregoing embodiment, the limitative torque characteristic T_P (N) with the number N of revolutions of the engine used as a variable therefor is set but the limitative torque of the pump may be fixedly set to a constant value T_{PA} as shown in Fig. 19. Incidentally, it is preferable that this limitative torque value T_{PA} is set to a value as large as possible on the assumption that an operation of the engine is not interrupted.

When an inclination angle of the swash plate in the pump is set so as to obtain the constant torque T_{PA} shown in Fig. 19 while the pressure sensor is abnormal in function, a torque of which intensity is represented by an inclined line in Fig. 20 can be absorbed by the pump.

When a series of processings are executed in the above-described manner, the pump outputs the torque T_P (N) or T_{PA} even when the pressure sensor is abnormal in function. Thus, for instance, in a case of a vehicle for which this pump is used as a power source for movement, it is possible to displace the vehicle to a repairing factory or the like.

Incidentally, in the foregoing embodiment, the characteristic T_P shown in Fig. 18 is stored in the memory and thereby it is possible to calculate a limitative torque value which conforms to T_P (N) with reference to N.

Since an apparatus for controlling a hydraulic pump according to the present invention functions in the above-described manner, it is advantageous that the apparatus is applied to a hydraulic pump for a construction machine which has a need of reducing fuel consumption cost and increasing an operational efficiency of the pump.

Claims

1. An apparatus for controlling a variable displacement type hydraulic pump (2) including an engine (1) as a driving power source comprising:

   means (5) for detecting the number of revolution of said engine,

   means (6) for detecting a pressure (P) of hydraulic oil delivered from said hydraulic pump,

   means for setting a pump absorption torque characteristic which monotonously decreases with reference to the number of revolutions of the engine,

   means (3) for looking for an inclination angle of a swash plate (2a) in the pump (2) with reference to said pump absorption torque characteristic and said pressure of hydraulic oil delivered from the pump,

   means (8) for controlling said swash plate in the pump so as to assure said inclination angle of the swash plate, and

   means for reducing the number of revolutions of the engine (1) under a condition that the absorption torque absorbed by the pump (2) does not exceed a rated torque of the engine (1).

2. An apparatus for controlling a hydraulic pump (2) as claimed in claim 1, wherein said pump absorption torque characteristic is a torque characteristic with which a constant work is undertaken by said pump (2).

3. An apparatus for controlling a hydraulic pump (2) as claimed in claim 1 or 2, wherein said apparatus further includes means (12) for setting a plurality of kinds of pump absorption torque characteristics and means (13) for selecting said characteristics.

4. An apparatus for controlling a hydraulic pump (2) as claimed in one of claims 1-3, wherein said condition under which an absorption torque absorbed by the engine (1) does not exceed an allowable torque is set with reference to the number of revolution of the engine (1) in respect of a point where the pump absorption torque characteristic intersects a rated torque characteristic of the engine.

5. An apparatus for controlling a hydraulic pump (2) as claimed in one of claims 1-4, wherein said means

for reducing the number of revolutions of the engine (1) is so constructed that the number of revolutions of the engine (1) is reduced at a predetermined time interval by a very small number of revolutions.

6. An apparatus for controlling a variable displacement hydraulic pump (22) including an engine (21) as a driving power source, said engine (21) having a constant horsepower characteristic at a predetermined revolution range, comprising;

means (23) for detecting the number of revolutions of the engine,

means (26) for setting a target pump absorption torque (T_P) in accordance with the following equation,

$$T_P = T_E (N_c) + K (N - N_c)$$

where

$T_E (N_c)$; rated torque characteristic of the engine at a predetermined revolution number range
K; constant
N; number of revolutions of the engine
N_c; target number of revolutions of the engine

means for controlling a swash plate (22a) in the pump (22) so as to obtain an absorption torque with reference to said target torque absorption torque and a pressure of hydraulic oil delivered from the pump (22), and

means (26) for reducing said target number of revolutions of the engine under a condition that the inclination angle of said swash plate (22a) is smaller than a preset angle (θ).

7. An apparatus for controlling a hydraulic pump as claimed in claim 6, wherein said means (26) for reducing the target number of revolutions of the engine is so constructed that the number of revolutions of the engine is reduced at a predetermined time interval by a very small number of revolutions.

8. An apparatus for controlling a hydraulic pump (2,22,32) as claimed in one of claims 1-7, wherein said apparatus further comprises:

means for detecting an abnormality with said pressure detecting means (6,36),

means for setting a pump absorption torque characteristic so as to reduce an absorption torque absorbed by the pump (2,22,32) lower than an output torque from the engine, and

means for controlling said inclination angle of said swash plate (2a,22a,32a) in the pump (2,22,32) so as to allow said absorption torque absorbed by the pump to exhibit a value which conforms to said pump absorption torque characteristic when said means (6,36) for detecting a pressure of hydraulic oil delivered from the pump becomes abnormal in function.

9. An apparatus for controlling a hydraulic pump (2,22,32) as claimed in claim 8, wherein said pump absorption torque characteristic is represented by a function which varies in dependence on the number of revolutions of the engine.

10. An apparatus for controlling a hydraulic pump (2,22,32) as claimed in claim 8, wherein said pump absorption torque characteristic is a characteristic which exhibits a constant value relative to the number of revolutions of the engine.

Revendications

1. Une unité de commande d'une pompe hydraulique (2) du type à déplacement variable incluant un moteur (1) comme source d'énergie d'entraînement comprenant :
   un moyen (5) de détection de la vitesse de rotation dudit moteur,
   un moyen (6) de détection d'une pression d'huile hydraulique (P) refoulée par ladite pompe hydraulique,

un moyen d'établissement d'une caractéristique de couple d'absorption de pompe qui diminue de façon monotone par rapport à la vitesse de rotation du moteur,

un moyen (3) de recherche d'un angle d'inclinaison d'un plateau oscillant (2a) dans ladite pompe (2) en fonction de ladite caractéristique de couple d'absorption de pompe et de ladite pression d'huile hydraulique refoulée par la pompe,

un moyen (8) de commande dudit plateau oscillant dans la pompe de façon à assurer ledit angle d'inclinaison du plateau oscillant, et

un moyen de réduction de la vitesse de rotation du moteur (1) dans une condition où le couple d'absorption absorbé par la pompe (2) ne dépasse pas un couple nominal du moteur (1).

2. Une unité de commande d'une pompe hydraulique (2) selon la revendication 1, dans laquelle ladite caractéristique de couple d'absorption de pompe est une caractéristique de couple au moyen de laquelle un travail constant est effectué par ladite pompe (2).

3. Une unité de commande d'une pompe hydraulique (2) selon la revendication 1 ou 2, dans laquelle ladite unité comprend en outre un moyen (12) pour établir plusieurs types de caractéristiques de couple d'absorption de pompe et un moyen (13) pour choisir lesdites caractéristiques.

4. Une unité de commande d'une pompe hydraulique (2) selon l'une des revendications 1 à 3, dans laquelle ladite condition dans laquelle un couple d'absorption absorbé par le moteur (1) ne dépasse pas un couple admissible est établie en fonction de la vitesse de rotation du moteur (1) selon un point où la caractéristique de couple d'absorption de pompe est intersectée par une caractéristique de couple nominal du moteur.

5. Une unité de commande d'une pompe hydraulique (2) selon l'une des revendications 1 à 4, dans laquelle ledit moyen de réduction de la vitesse de rotation du moteur (1) est d'une structure telle que la vitesse de rotation du moteur (1) est réduite d'une très faible valeur de vitesse de rotation à un intervalle de temps prédéterminé.

6. Une unité de commande d'une pompe hydraulique (22) à déplacement variable comprenant un moteur (21) comme source d'énergie d'entraînement, ledit moteur (1) présentant une caractéristique de puissance constante à une plage de vitesse de rotation prédéterminée, comprenant :

un moyen (23) pour détecter la vitesse de rotation du moteur,

un moyen (26) pour établir un couple cible d'absorption de pompe (T_P) conformément à l'équation suivante

$$T_P = T_E (N_c) + K (N - N_c)$$

où $T_E (N_c)$ = caractéristique de couple nominal du moteur à une plage de vitesse de rotation prédéterminée

K = constante

N = vitesse de rotation du moteur

N_c = vitesse cible de rotation du moteur

un moyen de commande d'un plateau oscillant (22a) dans la pompe (22) de façon à obtenir un couple d'absorption en fonction dudit couple cible d'absorption et d'une pression d'huile hydraulique refoulée par la pompe (22), et

un moyen (26) de réduction de ladite vitesse cible de rotation du moteur dans une condition où l'angle d'inclinaison dudit plateau oscillant (22a) est inférieur à un angle prédéterminé (θ).

7. Une unité de commande d'une pompe hydraulique selon la revendication 6, dans laquelle ledit moyen (26) de réduction de la vitesse cible de rotation du moteur est d'une structure telle que la vitesse de rotation du moteur est réduite d'une très faible valeur de vitesse de rotation à un intervalle de temps prédéterminé.

8. Une unité de commande d'une pompe hydraulique (2, 22, 32) selon l'une des revendications 1 à 7, dans laquelle ladite unité comprend en outre :

un moyen de détecter une anomalie concernant ledit moyen détecteur de pression (6, 36),

un moyen d'établissement d'une caractéristique de couple d'absorption de pompe de façon à

réduire un couple d'absorption absorbé par la pompe (2, 22, 32) en dessous d'un couple de sortie du moteur, et

un moyen de commande dudit angle d'inclinaison dudit plateau oscillant (2a, 22a, 32a) dans la pompe (2, 22, 32) de façon à permettre audit couple d'absorption absorbé par la pompe de présenter une valeur qui est conforme à ladite caractéristique de couple d'absorption de pompe lorsque le fonctionnement dudit moyen (6, 36) de détection de pression d'huile hydraulique refoulée par la pompe devient anormal.

9. Une unité de commande d'une pompe hydraulique (2, 22, 32) selon la revendication 8, dans laquelle ladite caractéristique de couple d'absorption de pompe est représentée par une fonction qui varie selon la vitesse de rotation du moteur.

10. Une unité de commande d'une pompe hydraulique (2, 22, 32) selon la revendication 8, dans laquelle ladite caractéristique de couple d'absorption de pompe est une caractéristique qui présente une valeur constante par rapport à la vitesse de rotation du moteur.

Patentansprüche

1. Vorrichtung zum Steuern einer variablen hydraulischen Verdrängungspumpe (2), die einen Motor (1) als Antriebskraftquelle aufweist, mit:
   - einer Einrichtung (5) zum Ermitteln der Drehzahl des Motors,
   - einer Einrichtung (6) zum Ermitteln des Drucks (P) des von der Hydraulikpumpe geförderten Hydrauliköls,
   - einer Einrichtung zum Einstellen einer Absorptionsdrehmomentkennlinie der Pumpe, die mit der Drehzahl des Motors monoton abnimmt,
   - einer Einrichtung (3) zum Ermitteln nach des Neigungswinkels einer Taumelscheibe (2a) in der Pumpe (2) in bezug auf die Absorptionsdrehmomentkennlinie der Pumpe und den Druck des von der Pumpe gelieferten Hydrauliköls,
   - einer Einrichtung (8) zum Steuern der Taumelscheibe in der Pumpe derart, daß der genannte Neigungswinkel der Taumelscheibe eingenommen wird, und
   - einer Einrichtung zum Verringern der Drehzahl des Motors (1) unter der Bedingung, daß das von der Pumpe (2) absorbierte Absorptionsdrehmoment ein Nenndrehmoment des Motors (1) nicht übersteigt.

2. Vorrichtung zum Steuern einer Hydraulikpumpe (2) nach Anspruch 1, bei der die Absorptionsdrehmomentkennlinie der Pumpe eine Pumpenkennlinie ist, bei der die Pumpe (2) eine konstante Arbeit verrichtet.

3. Vorrichtung zum Steuern einer Hydraulikpumpe (2) nach Anspruch 1 oder 2, wobei die Vorrichtung ferner eine Einrichtung (12) zum Einstellen mehrerer Arten von Absorptionsdrehmomentkennlinien der Pumpe und eine Einrichtung (13) zum Auswählen der Kennlinien aufweist.

4. Vorrichtung zum Steuern einer Hydraulikpumpe (2) nach einem der Ansprüche 1 - 3, bei der ie Bedingung, unter der ein von dem Motor (1) absorbiertes Absorptionsdrehmoment ein zulässiges Drehmoment nicht übersteigt, in bezug auf die Drehzahl des Motors (1) auf einen Punkt eingestellt ist, in dem die Absorptionsdrehmomentkennlinie der Pumpe eine Nenndrehmomentkennlinie des Motors schneidet.

5. Vorrichtung zum Steuern einer Hydraulikpumpe (2) nach einem der Ansprüche 1 - 4, bei der die Einrichtung zur Verringerung der Drehzahl des Motors (1) derart konstruiert ist, daß die Drehzahl des Motors (1) in einem vorbestimmten Zeitintervall um eine sehr geringe Anzahl von Umdrehungen verringert wird.

6. Vorrichtung zum Steuern einer variablen hydraulischen Verdrängungspumpe (22) mit einem Motor (21) als Antriebskraftquelle, wobei der Motor (21) in einem vorbestimmten Drehzahlbereich eine konstante Leistungskennlinie aufweist, mit:
   - einer Einrichtung (23) zum Ermitteln der Drehzahl de Motors,
   - einer Einrichtung zum Einstellen eines Ziel-Absorptionsdrehmont (T_P) der Pumpe nach der

folgenden Gleichung

$TP = T_E (N_C) + K (N - N_C)$

wobei $T_E (N_C)$ die Nenn-Drehmomentkennlinie in einem vorbestimmten Drehzahlbereich ist;

K eine Konstante ist;

N die Drehzahl des Motors ist;

N_C die Ziel-Drehzahl des Motors ist;

- einer Einrichtung zum Steuern einer Taumelscheibe (22a) in der Pumpe (22), derart, daß ein Absorptionsdrehmoment in bezug auf das Ziel-Drehmomentabsorptionsdrehmoment und einen Druck des von der Pumpe (22) geförderten Hydrauliköls erhalten wird, und
- einer Einrichtung (26) zum Verringern der Ziel-Drehzahl des Motors unter der Bedingung, daß der Neigungswinkel der Taumelscheibe (22a) kleiner als ein voreingestellter Winkel (θ) ist.

7. Vorrichtung zum Steuern einer Hydraulikpumpe nach Anspruch 6, bei der die Einrichtung (26) zum Verringern der Ziel-Drehzahl des Motors derart konstruiert ist, daß die Drehzahl des Motors in einem vorbestimmten Zeitintervall um eine sehr geringe Anzahl von Umdrehungen verringert wird.

8. Vorrichtung zum Steuern einer Hydraulikpumpe (2, 22, 32) nach einem der Ansprüche 1-7, wobei die Vorrichtung ferner aufweist:
   - eine Einrichtung zum Erkennen einer Störung der Druckermittlungseinrichtung (6, 36),
   - einer Einrichtung zum Einstellen einer Absorptionsdrehmomentkennlinie der Pumpe derart, daß ein von der Pumpe (2, 22, 32) absorbiertes Absorptionsdrehmoment geringer wird als ein Ausgangsdrehmoment des Motors, und
   - einer Einrichtung zum Steuern des Neigungswinkles der Taumelscheibe (2a, 22a, 32a) in der Pumpe (2, 22, 32) derart, daß das von der Pumpe absorbierte Absorptionsdrehmoment einen Wert hat, der mit der Absorptionsdrehmomentkennlinie übereinstimmt, wenn die Einrichtung (6, 36) zum Ermitteln des Drucks des von der Pumpe geförderten Hydrauliköls einen abnormalen Zustand annimmt.

9. Vorrichtung zum Steuern einer Hydraulikpumpe (2, 22, 32) nach Anspruch 8, bei der die Absorptionsdrehmomentkennlinie der Pumpe durch eine Funktion repräsentiert ist, die in Abhängigkeit von der Drehzahl des Motors variiert.

10. Vorrichtung zum Steuern einer Hydraulikpumpe (2, 22, 32) nach Anspruch 8, bei der die Absorptionsdrehmomentkennlinie der Pumpe eine Kennlinie ist, die relativ zur Drehzahl des Motors einen konstanten Wert hat.

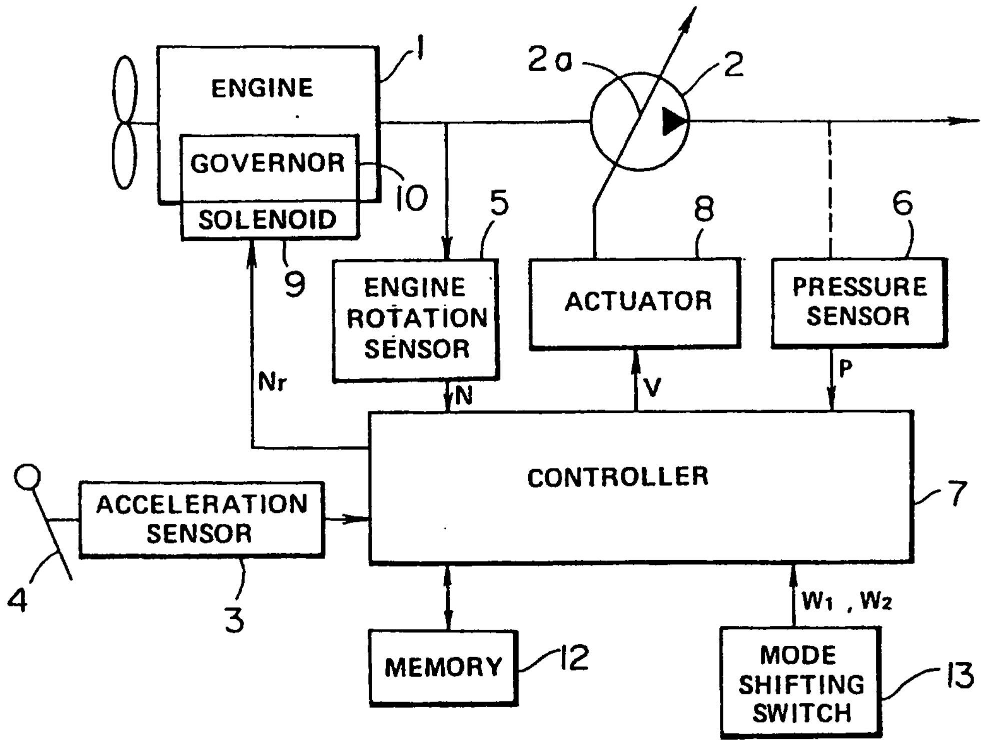

ENGINE
GOVERNOR
SOLENOID
ENGINE ROTATION SENSOR
ACTUATOR
PRESSURE SENSOR
CONTROLLER
ACCELERATION SENSOR
MEMORY
MODE SHIFTING SWITCH
Nr
N
V
P
W1 , W2
1
2
2a
3
4
5
6
7
8
9
10
12
13

FIG.1

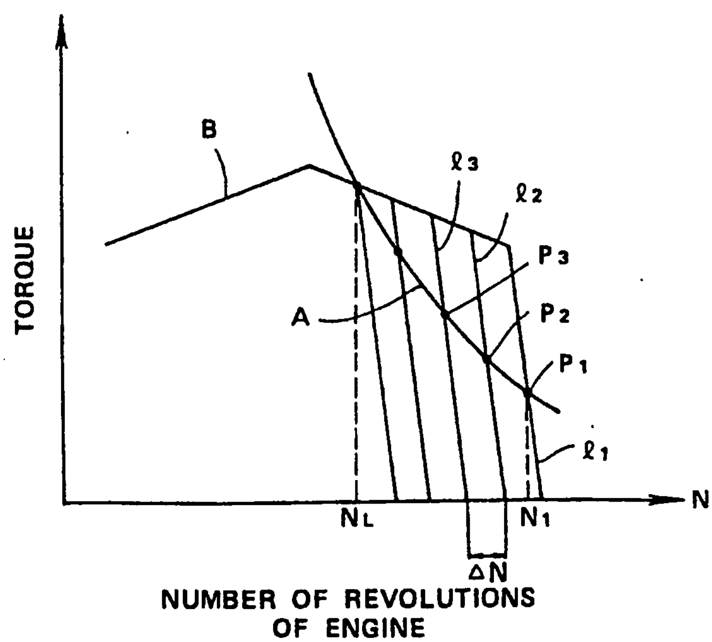

TORQUE
B
A
ℓ3
ℓ2
ℓ1
P3
P2
P1
NL
N1
ΔN
N
NUMBER OF REVOLUTIONS OF ENGINE

FIG.3

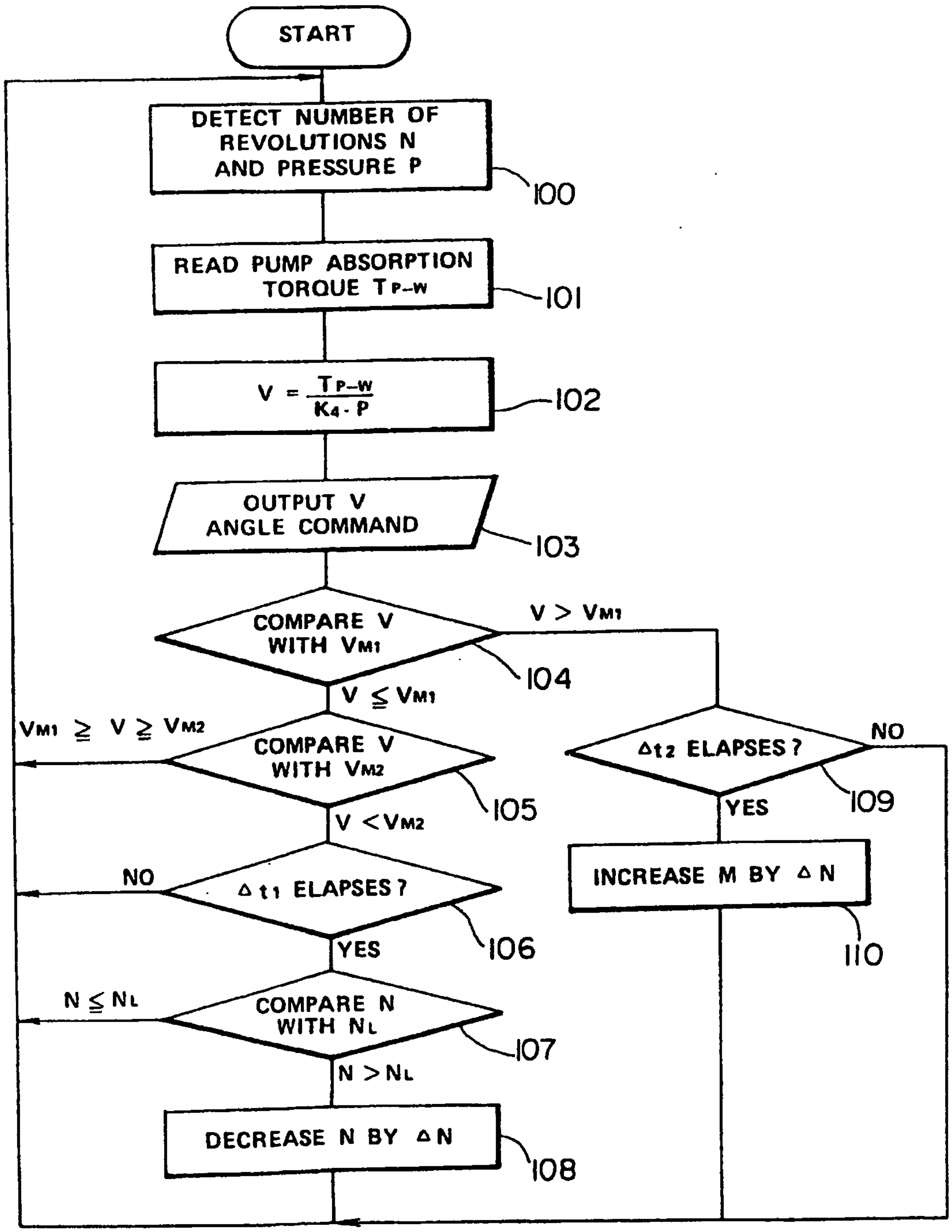

START
DETECT NUMBER OF REVOLUTIONS N AND PRESSURE P
100
READ PUMP ABSORPTION TORQUE T_{P-W}
101
$V = \frac{T_{P-W}}{K_4 \cdot P}$
102
OUTPUT V ANGLE COMMAND
103
COMPARE V WITH V_{M1}
104
$V > V_{M1}$
$V \leqq V_{M1}$
$V_{M1} \geqq V \geqq V_{M2}$
COMPARE V WITH V_{M2}
105
$V < V_{M2}$
NO
Δt_1 ELAPSES ?
106
YES
$N \leqq N_L$
COMPARE N WITH N_L
107
$N > N_L$
DECREASE N BY ΔN
108
Δt_2 ELAPSES ?
NO
109
YES
INCREASE M BY ΔN
110

FIG.2

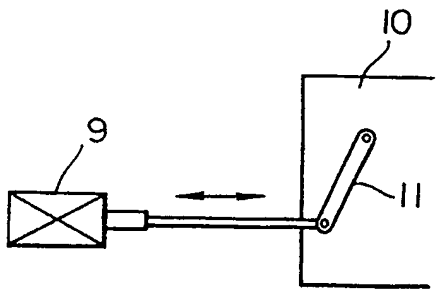


FIG. 4

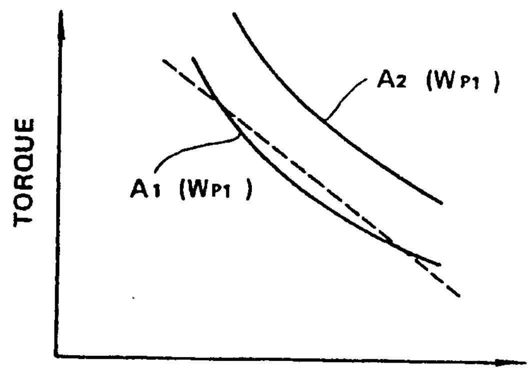


FIG. 5

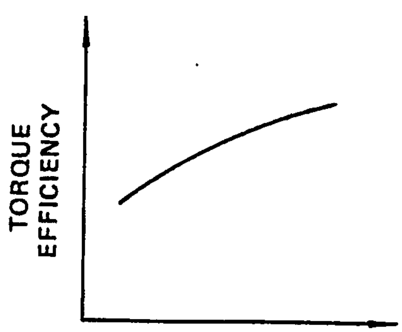


FIG. 6

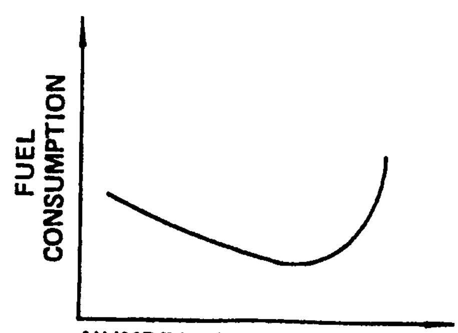
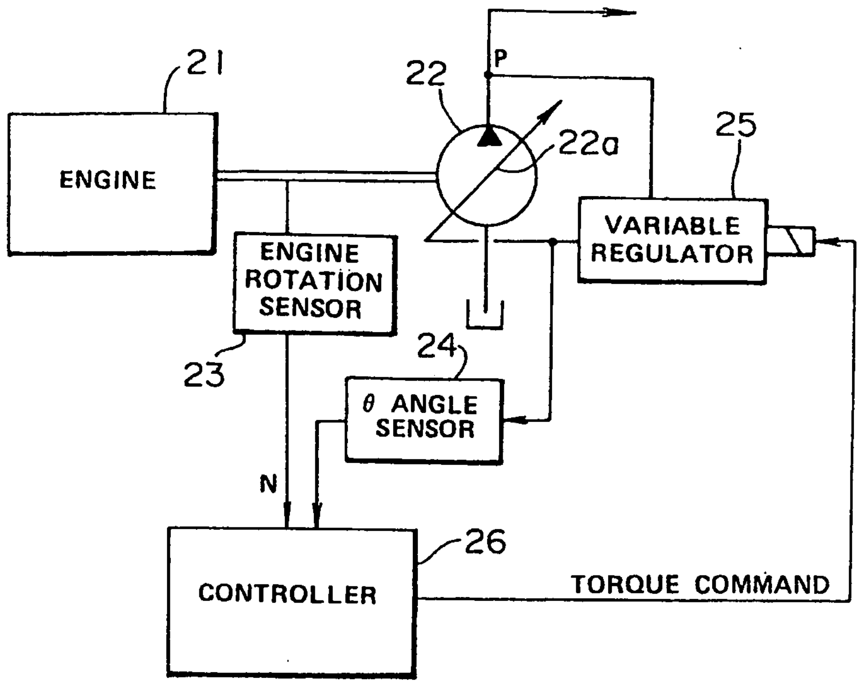
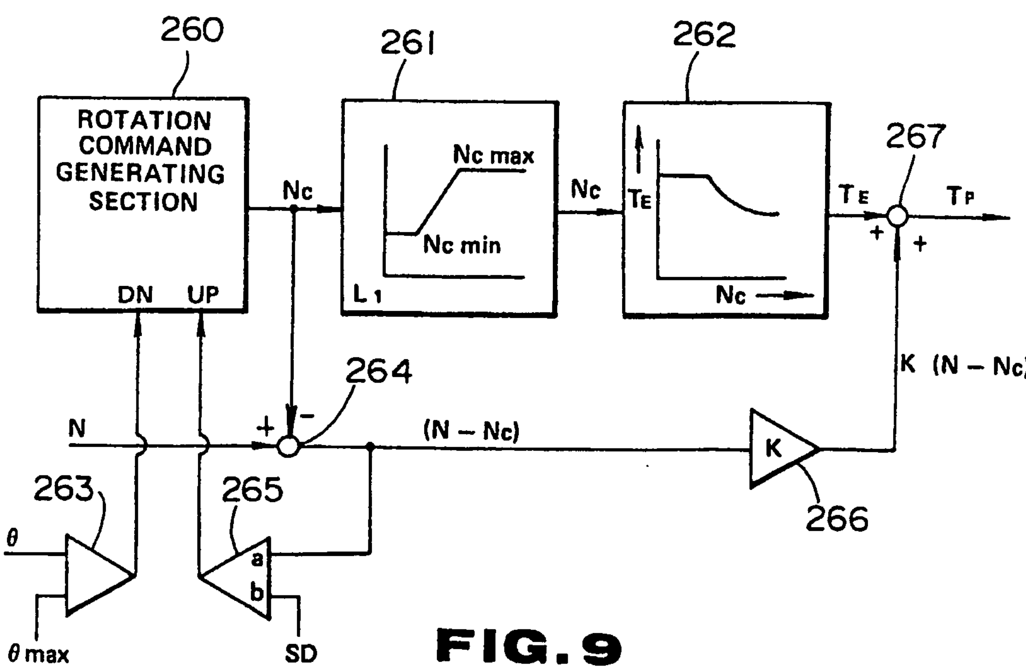


FIG. 7



21
22
P
22a
25
ENGINE
VARIABLE REGULATOR
ENGINE ROTATION SENSOR
23
24
θ ANGLE SENSOR
N
26
CONTROLLER
TORQUE COMMAND

FIG. 8



260
261
262
ROTATION COMMAND GENERATING SECTION
Nc
Nc max
Nc min
L1
Nc
TE
Nc
TE
TP
267
+
+
DN
UP
K (N – Nc)
264
N
+
-
(N – Nc)
K
263
265
266
θ
a
b
θ max
SD

FIG. 9

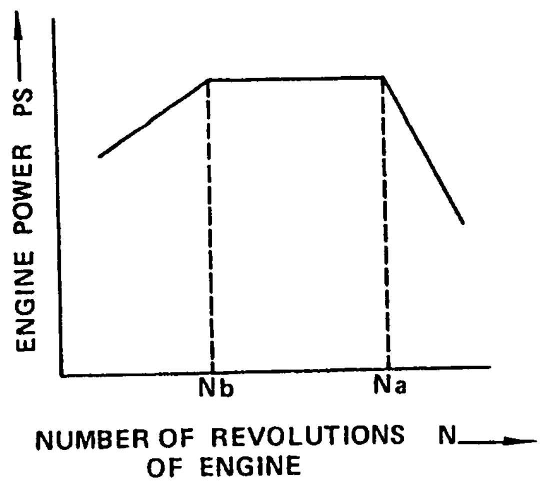


FIG.10

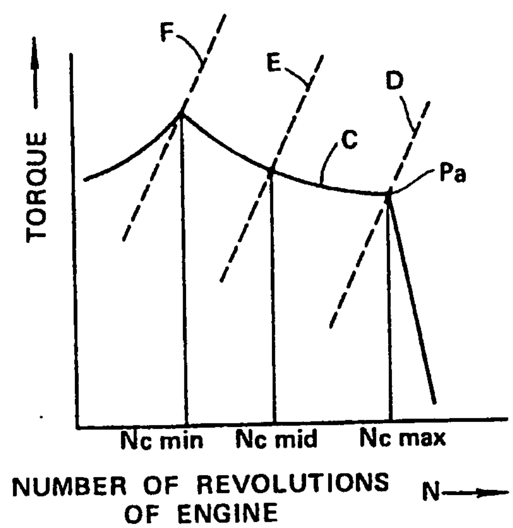


FIG.11

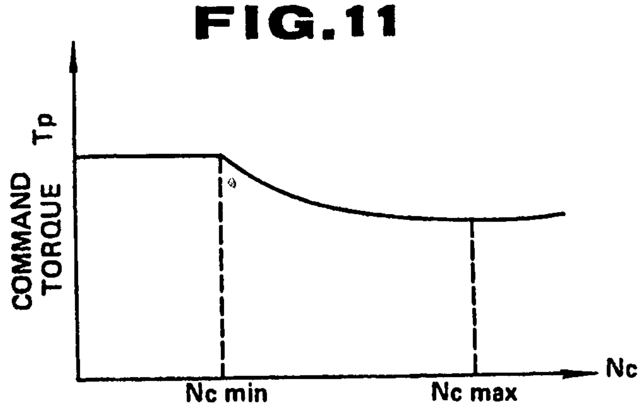


FIG.12

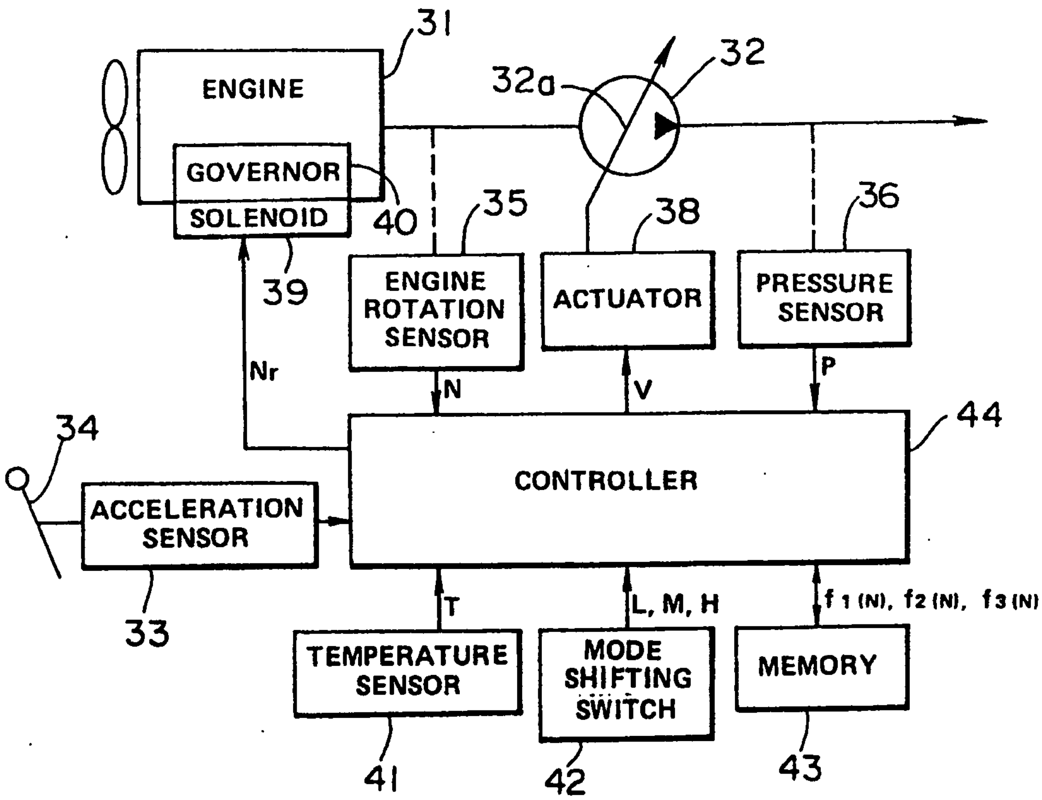

31
ENGINE
GOVERNOR
SOLENOID
40
39
32a
32
35
ENGINE ROTATION SENSOR
38
ACTUATOR
36
PRESSURE SENSOR
Nr
N
V
P
44
CONTROLLER
34
ACCELERATION SENSOR
33
T
L, M, H
f 1 (N), f2 (N), f3 (N)
TEMPERATURE SENSOR
MODE SHIFTING SWITCH
MEMORY
41
42
43

FIG.13

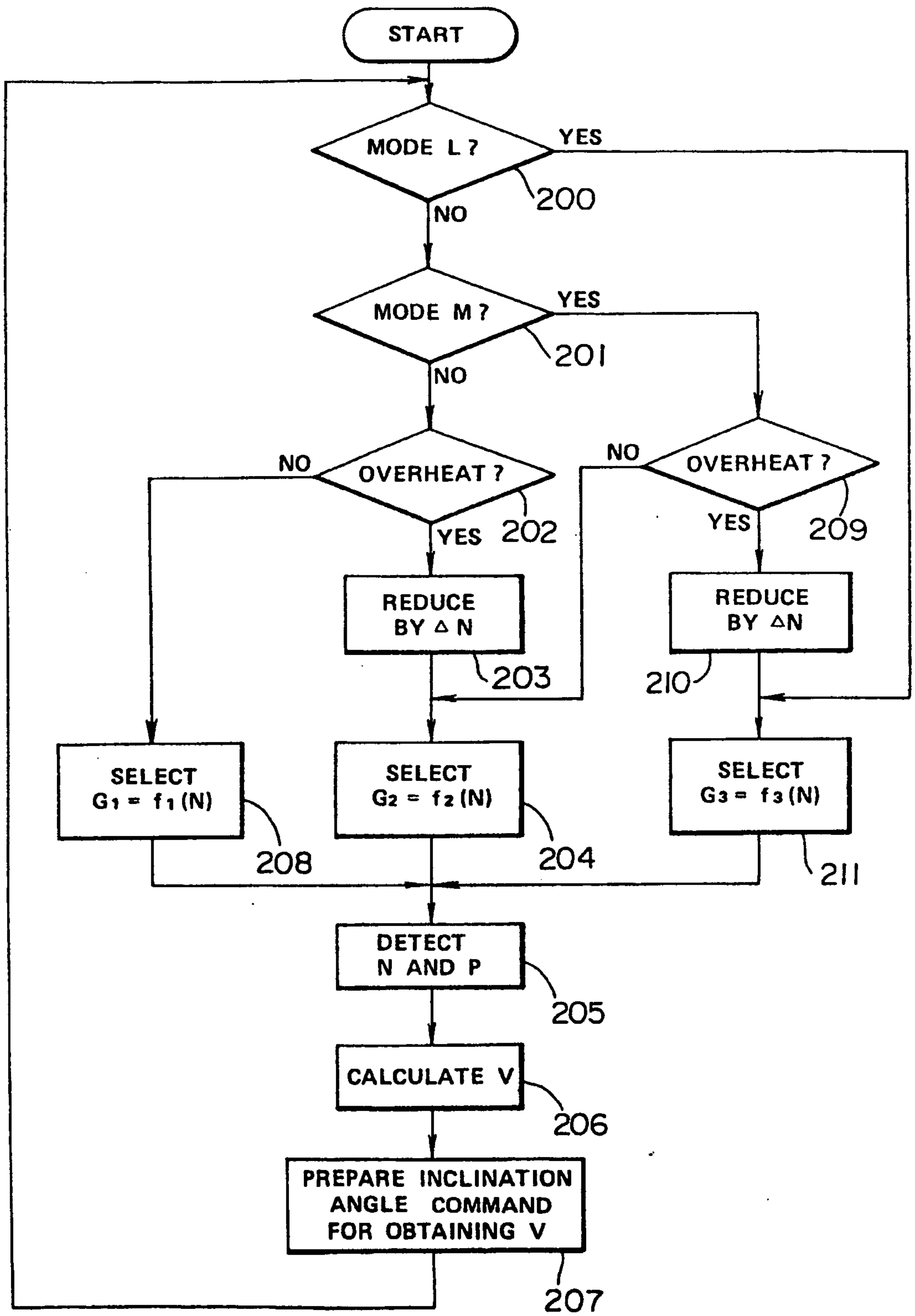

START
MODE L ?
YES
NO
200
MODE M ?
YES
NO
201
NO
OVERHEAT ?
YES
202
NO
OVERHEAT ?
YES
209
REDUCE BY Δ N
203
REDUCE BY ΔN
210
SELECT G1 = f1 (N)
208
SELECT G2 = f2 (N)
204
SELECT G3 = f3 (N)
211
DETECT N AND P
205
CALCULATE V
206
PREPARE INCLINATION ANGLE COMMAND FOR OBTAINING V
207

FIG.14

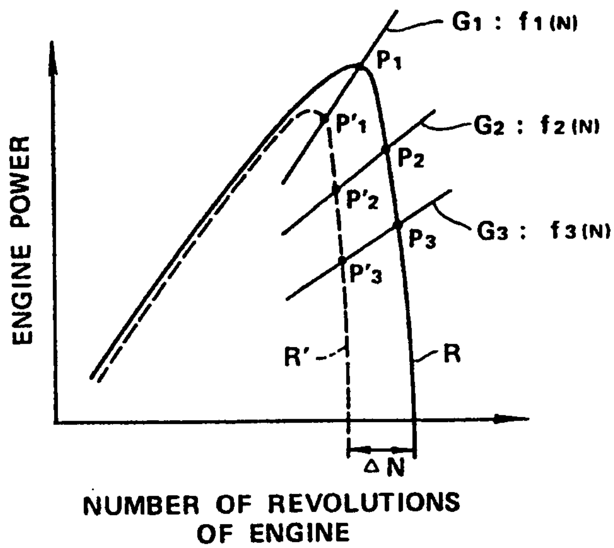

G1 : f1 (N)
P1
P'1
G2 : f2 (N)
P2
P'2
G3 : f3 (N)
P3
P'3
ENGINE POWER
R'
R
ΔN
NUMBER OF REVOLUTIONS
OF ENGINE

FIG.15

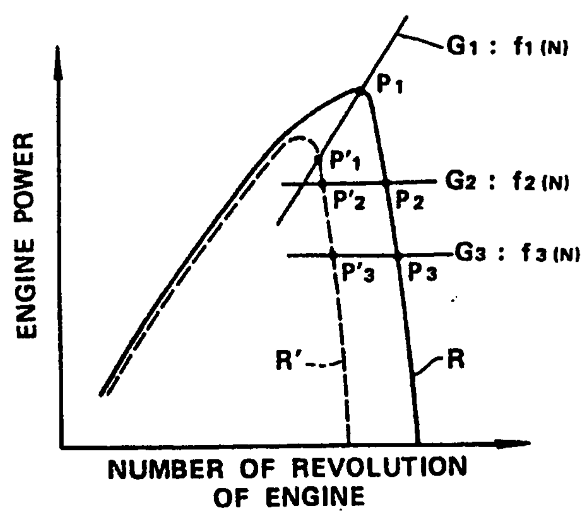

G1 : f1 (N)
P1
P'1
G2 : f2 (N)
P'2
P2
G3 : f3 (N)
P'3
P3
ENGINE POWER
R'
R
NUMBER OF REVOLUTION
OF ENGINE

FIG.16

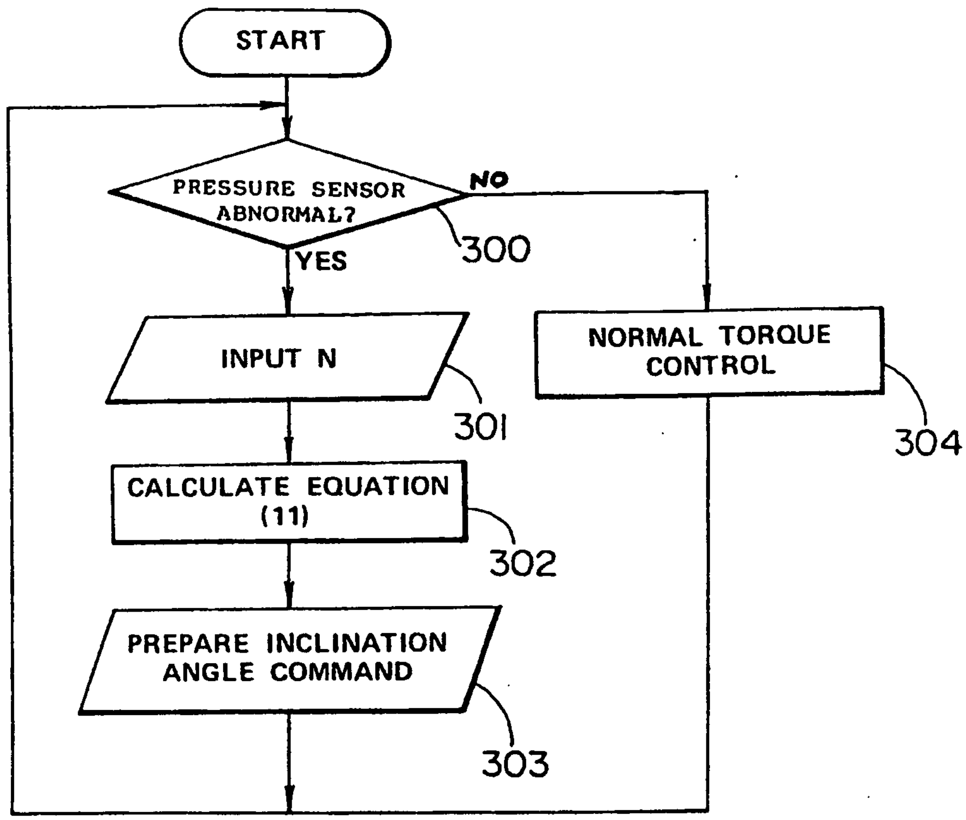

START
PRESSURE SENSOR ABNORMAL?
NO
YES
300
INPUT N
301
CALCULATE EQUATION (11)
302
PREPARE INCLINATION ANGLE COMMAND
303
NORMAL TORQUE CONTROL
304

FIG.17

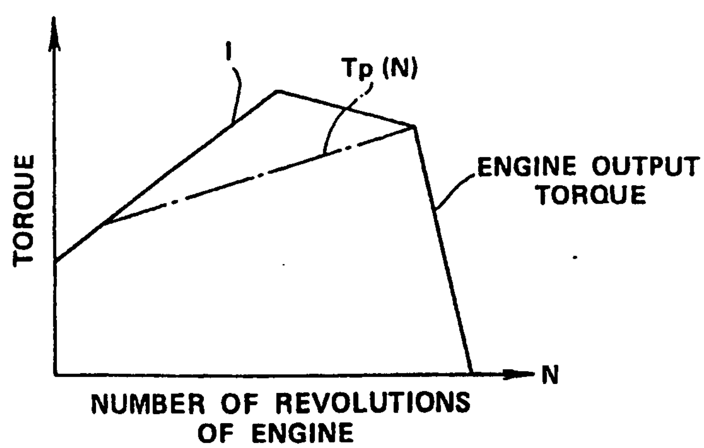

I
Tp (N)
ENGINE OUTPUT TORQUE
TORQUE
N
NUMBER OF REVOLUTIONS OF ENGINE

FIG.18

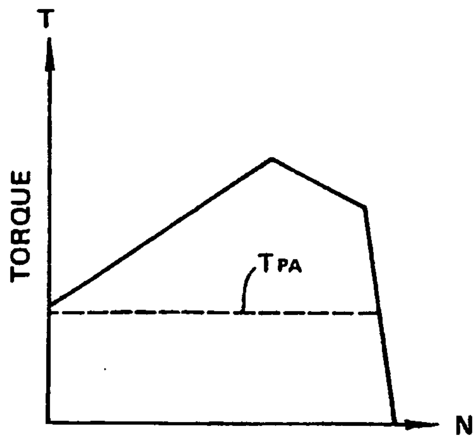


FIG.19

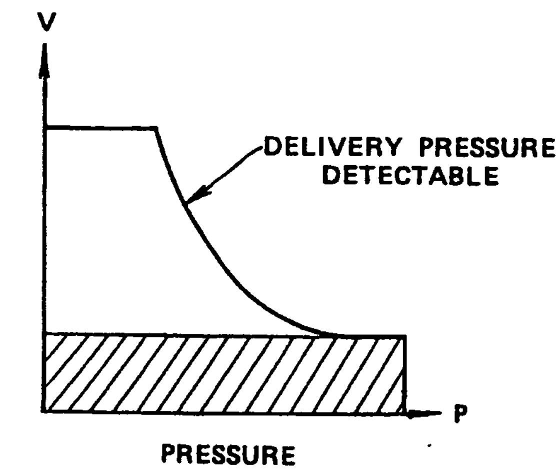


FIG.20